REPORT No. 106

TURBULENCE IN THE AIR TUBES OF RADIATORS FOR AIRCRAFT ENGINES

By S. R. PARSONS
Bureau of Standards

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This report describes an investigation of the characteristics of flow in the air passages of aircraft radiators, the work being done at the Bureau of Standards for the National Advisory Committee for Aeronautics.

RÉSUMÉ.

The existence of turbulent flow in the air passages of aircraft radiators, and of variations in character or degree of turbulence with different types of construction, is shown by the following experimental evidence:

(1) Pressure gradients along the air tubes are roughly proportional to the 1.7 power of the speed, which is characteristic of turbulent flow in long circular tubes of the same diameters.

(2) The surface cooling coefficients of radiators vary widely (0.002 to 0.007) when expressed as heat dissipated per unit time, per unit cooling surface, per unit temperature difference between air and water, and at a given average linear speed through the tubes.

(3) A fine wire electrically heated shows different cooling coefficients in the air tubes of different radiators.

(4) Temperature gradients in the air tubes are of the form characteristic of turbulent flow, and fail to show sudden breaks, such as might indicate a dividing line between regions of viscous and of turbulent flow.

The use of special devices for increasing turbulence may increase the heat transfer per unit surface for a given flow of air through the radiator, but decreases that flow for a given speed of flight, and increases head resistance. At very low flying speeds, or in cases where the radiator is mounted in the nose of the fuselage, turbulence devices may sometimes be used to advantage, but every type known to this bureau is detrimental to the general performance of the radiator at high speeds.

INTRODUCTION.

The primary requirement of a cooling radiator is evidently that it shall dissipate heat; and for cooling the engines of aircraft it is very important that the head resistance shall be low. But both heat transfer and head resistance are greatly affected not only by the speed of air past the cooling surfaces, but by the character of the flow—whether the air passes through the radiator in smooth streams, or with eddies and vortices. Furthermore, if the flow is turbulent, the questions arise whether the turbulence can be increased by changes in construction, and if so whether the result is beneficial or harmful to the general performance of the radiator.

This paper will not attempt to take up the questions of turbulence from a theoretical standpoint, but rather to present experimental evidence bearing on the problems and to state certain conclusions that seem to be warranted by the evidence. In particular, the paper seeks to answer the questions (1) whether the flow of air through ordinary radiators is turbulent at usual speeds; (2) whether there may be different degrees or types of turbulence; and (3) whether forms of construction intended to increase turbulence are beneficial or detrimental to the radiator.

The heat dissipated by a radiator is taken up by streams of air flowing through its tubes, and both the quantity of air delivered by the streams, and their condition of turbulence, are important factors in the dissipation of heat. Since air transmits heat only slowly by conduction, but principally by convection, the amount of heat taken from the metal surface depends very greatly on the number of *different* molecules of air that come in contact with the metal, and the most rapid transfer of heat requires a considerable scouring of the surface, while a layer of stagnant air acts as an effective insulator. But the collision of molecules of air with those of metal, imparting heat to the air because of the molecular motion in the hot metal, also ends because of the *mass* motion, to drag the air along with the radiator, instead of allowing it to pass through the tubes; and while turbulence in the air streams facilitates heat transfer, it also increases head resistance.

It is well known that in long tubes with smooth walls, at points not too near the ends, air flow is of two kinds, depending upon the relations between diameter of the tube and speed, viscosity and density of air. At low speeds, the flow is practically along stream lines parallel to the walls of the tubes, and is called viscous or "streamline" flow; while at higher speeds in the same tube, the flow is broken up into vortices, and is called turbulent flow. In viscous flow the skin friction, or resistance to flow, is due principally to the viscosity of the air, and is roughly proportional to the first power of the speed; while in turbulent flow viscosity is of less importance than density, and the resistance is roughly proportional to the square of the speed. The shortness of the tubes of radiators, and the irregular and broken forms often employed, make it unsafe to apply the theory of long tubes.

Most investigators measuring pressure drop in long tubes take no measurements of pressure nearer than 50 to 100 diameters to either end, but the total length of radiator tubes is usually not more than 10 to 30 diameters, and the fact that the rate of flow is ordinarily far above the critical velocity for a long tube of the same diameter is not sufficient basis for a statement that the flow in the radiator is turbulent.

It may be expected, however, that in a cluster of tubes such as a radiator, conditions corresponding to *long* tubes may be found much nearer the ends than in a single tube. Unless some kind of a mouthpiece is provided, air coming over the edges of a single tube enters from many directions, and the same is true to some extent of radiator tubes that are near the edges of the section; but the air that enters tubes near the center is confined by that entering the other tubes, and is fairly well directed even before it reaches the tubes.

Three general methods may be used for detecting turbulence; a visual method using some kind of smoke; measurements of pressure gradients; and measurement of heat transfer or of temperature. The visual method has great advantages, but is inconvenient for work inside of the radiator because of difficulty of arranging the apparatus so that it shall not disturb the flow of air, and at the same time so that air currents in the radiator tubes may be distinguished from currents before and behind it. The fact that the pressure gradient along an air stream is roughly proportional to the first power of the speed for streamline or viscous flow, and to the square of the speed for turbulent flow, may be used to determine the nature of the motion. The transfer of heat from a surface swept by a stream of air depends upon the turbulence of the stream as well as upon its velocity, and while with the present limited knowledge of coefficients of heat transfer a single measurement might be of little value for detecting the presence or absence of turbulence, considerable information may be gained from comparative measurements. Temperature measurements at different points in a stream of air that is being heated or cooled may, if reliable, furnish some indication of the condition of the air, by showing how heat is transmitted through different portions of the stream.

The experimental work on which the conclusions are based was carried out in an 8-inch (20 cm) square wind tunnel, and the evidence here presented is not sufficient for a confident answer to the question whether the same kind of flow is found in a stationary radiator with air blowing through it, as in a radiator moving through still air. It seems to be shown conclusively, however, that at least when the radiator is in the wind tunnel there are characteristic conditions of turbulence in different types of core, and it seems reasonable to suppose that such characteristic conditions would also be found in radiators moving through still air.

The experimental work undertaken for the present investigation includes the following parts, which will be treated in detail:

I. Flow in the channel in front of the radiator.
II. Pressure gradients in radiator tubes.
III. Cooling coefficients of radiators.
IV. Cooling of wires in an air stream.
V. Temperature gradients.

I. FLOW IN THE CHANNEL IN FRONT OF THE RADIATOR.

The characteristics of the air stream in the tunnel before entering the radiator were studied by two methods—by measuring the velocity at different points across a section of the stream and by observing the behavior of ammonium chloride smoke.

The velocity was measured by a movable pitot-static tube, and found to be uniform within 2 per cent to within 1 cm ($\frac{1}{2}$ inch) of the walls.

Ammonium chloride smoke was introduced into the tunnel through a glass tube about 8 mm ($\frac{5}{16}$ inch) in diameter, projecting through the bell mouth of the tunnel to a few centimeters beyond the straightening honeycomb at its entrance. On looking into the tunnel, either downstream through the mouth or across the stream through a window in the top, the smoke was seen to follow a straight course down the stream, with very little spreading. When a radiator was placed in the tunnel the smoke was found deposited over an area of the face which was fairly sharply defined, rather than shading off gradually. At a distance of 1 meter from the mouth of the smoke tube, the areas ranged from 4 to 9 per cent of the cross section of the tunnel, indicating a slow mixing of the air stream.

II. PRESSURE GRADIENTS IN RADIATOR TUBES.

The measurement of static pressure and pressure gradients within the air tubes of radiators has been described in Technical Report No. 88, "Pressure Drop in Radiator Air Tubes," and subsequent to the preparation of that report other work has involved incidental measurements on a number of additional types of core. Figure 1, reproduced from Report No. 88, is typical of the results obtained by plotting pressures against distance along the tube at different speeds of air. In most cases measurements were taken at either three or four speeds, and the following table shows the powers of the speed to which the pressure gradients are proportional. In many cases the pressure curve has no straight portion, and the difference in pressure between two points inside of the tube was used in place of a gradient. The formula of C. H. Lees[1] for surface friction in long circular tubes gives an exponent of about 1.73 for the sizes found in most radiators, and the grouping of these powers around that value furnishes good evidence of turbulent flow.

TABLE I.—*Exponent of air flow to which pressure drop in the air tubes is proportional.*

Radiator.	Type.	Exponent.
A-7	Square cell	1.6
A-9	do	1.7
A-14	[illegible]	1.7
A-19	[illegible]	1.8
A-29	[illegible]	1.8
A-31	do	1.9
B-3	Pseudo-hexagonal	1.8
B-13	do	1.6
B-17	True hexagonal	1.7
C-2	Pseudo-circular	1.7
C-12	True circular cell	1.8
C-13	do	1.7
D-1	Irregular	1.6
D-3	do	1.9
D-4	do	1.5

[1] Proceedings Royal Society of London, A 91, 1914, p. 46.

III.—COOLING COEFFICIENTS OF RADIATORS.

Cooling coefficients of surfaces in radiators have been obtained from tests of heat transfer on about 60 types of core[2] and for the purpose of comparison have been reduced to heat dissipated per unit time per unit cooling surface, per unit temperature difference between the air and the water in the radiator, when the flow of air is such as to give a mean speed of 26.8 meters per second (60 mi./hr.) through the radiator tubes.[3] The effect of a large amount of indirect cooling surface (surface not backed by flowing water) is to decrease the value of the coefficient because of the fact that for a given temperature of *water*, indirect cooling surfaces have lower mean temperature than the water-tube walls. This effect seems wholly insufficient, however, to account for the wide variation of the coefficients shown in the following table:

TABLE II.—*Surface cooling coefficients of radiators.*

Coefficient in $\frac{\text{cal.}}{\text{sec.} \times \text{sq. cm.} \times {}^\circ\text{C.}}$

Number of radiators.	Type.	Dimensions of cell. Centimeters.	Dimensions of cell. Inches.	Coefficient × 1000.
	SQUARE CELLS.			
6	Smooth walls	0.6	1/4	2.7-3.1
2	Walls swaged	.6	1/4	2.8-2.9
3	....do	.8	5/16	2.6-2.8
9	Irregular, approximately square	.6	1/4	2.6-3.5
1	....do	1.2	1/2	2.4
	HEXAGONAL CELLS.			
12	Pseudo-cellular	.9	3/8	2.3-3.2
1	....do	.6	1/4	3.4
3	True hexagonal cells	.8	5/16	2.8-2.9
	CIRCULAR CELLS.			
1	Pseudo-cellular	1.2	1/2	2.2
3	True circular cells	.8	5/16	2.8-2.9
4	....do	.9	3/8	2.9-3.1
	OTHER FORMS.			
5	Irregular cells, smooth walls	....	....	2.1-3.4
3	Flat plate water tubes	....	....	2.9-3.1
4	Perforated plate water tubes (whistling type)	....	....	3.0-7.7
2	Spiral vanes, in good thermal contact	....	....	4.3-4.4

The table shows that for radiators whose air passages have straight smooth walls, the coefficient ranges from 0.0021 to 0.0034; for cells with broken walls ("pseudo-cellular" types), from 0.0024 to 0.0034; for the perforated plate types that whistle in an air stream, from 0.0030 to 0.0077; and for a type with spiral vanes, is about 0.0044. In general, the coefficient decreases as the size of the cell increases.

The wide range of coefficients even for straight tubes, the high coefficient for the section with spiral vanes, and the very high values found with some of the perforated plate types, appear to show very strong evidence of varying conditions of turbulence in the different classes. The presence of a peculiar turbulence in the perforated plate types is also indicated by their whistle.

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[2] Technical Report No. 63.

[3] This coefficient is the factor q of an empirical equation that has been found applicable to radiators with smooth straight tubes:

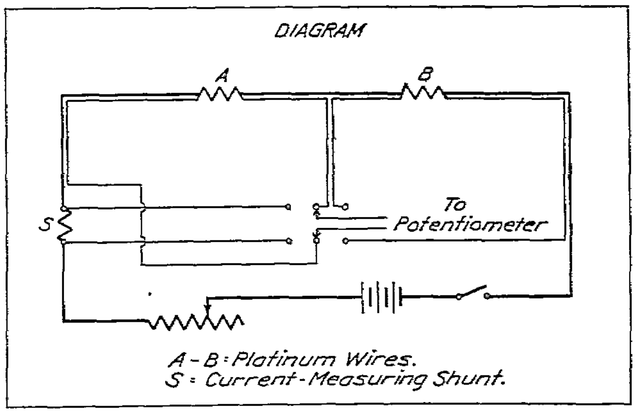
$$H = CMT\left(1 - e^{-\frac{qpx}{CM}}\right)$$

where H=heat transfer, units of power per unit frontal area.
C=specific heat of air at constant pressure.
M=air flow through the radiator, units of mass per unit time per unit frontal area.
T=difference between mean water temperature and temperature of air at entrance to radiator.
e=base of Naperian logarithms.
q=cooling coefficient, units of heat per unit time per unit surface per unit temperature difference.
p=total perimeter of air tubes (in frontal area) around a section perpendicular to the direction of air flow.
x=depth of radiator.

Coefficients for the radiators with true circular air tubes, if computed by the equation of Nusselt[4] would be 0.0026 which is somewhat lower than those observed. Nusselt's equation, however, can not be expected to apply to the radiator tubes, for it represents conditions with turbulent flow, in the central portions of *long* tubes.

IV. COOLING OF WIRES IN AN AIR STREAM.

The temperatures maintained by electrically heated wires, connected in series so that each should carry the same current, were used for a comparison of the cooling effect of the small streams of air flowing through a cold radiator, with that of the stream in the tunnel in front of it. A detailed description of the method follows:



Two pieces of 0.1 mm (0.004 inch) platinum wire, each about 6 cm (2.4 inches) long, and separated by 48 cm (19 inches) of No. 36 copper wire, were strung by copper leads straight along the channel so that one wire was in the stream some distance in front of the radiator, while the other was in one of the tubes, well back toward the rear face. The platinum wires, connected in series with a current-measuring shunt, as shown in the diagram, were heated by a small electric current, and potential drop was measured with a potentiometer across each platinum wire and the shunt. In order that the wires might disturb the air as little as possible, the potential and current leads were laid side by side (not twisted) and shellacked together for some distance from the platinum wires. The current leads supporting the wires were passed over rods set across the channel about 55 cm (22 inches) in front of and behind the radiator, and out through small holes in the floor of the tunnel, beyond which one wire was made fast, while a small weight hung on the other served to keep the wires taut.

The magnitude of the electric current was determined from the known resistance of the shunt and the potential drop across it, while from the current and the potential drops across the platinum wires, their resistances were obtained. The resistances when unheated were measured with small currents (about 0.006 amperes) and in an air stream of about 13 meters per second (30 miles per hour). It seemed hardly worth while to attempt to use the wires as resistance thermometers to the extent of measuring the temperatures to which they rose, but since for small temperature changes the resistance of the platinum wire is roughly proportional to the absolute temperature, the fractional increase in resistance of each wire was used as a rough measure of its rise in temperature on being heated.

Heat transfer from fine wires has been found to be proportional to the temperature difference and the square root of the velocity, for a silver wire in a tube of flowing water,[5] and for a platinum wire moved through air on a whirling arm,[6] and if the same relation is assumed

[4] Zeitschrift des Vereines Deutscher Ingenieure, 54, II, 1910, p. 1154.
[5] Rogovsky, Comptes Rendus, 136, June 8, 1903, p. 1391.
[6] King, Philosophical Transactions of the Royal Society of London, A 214, 1914, p. 373.

for the present case, a factor depending upon turbulence may be found as follows. The relation may be expressed by the equation

$$H = c\Theta\sqrt{v} \tag{1}$$

where

H = heat transfer from the wire, in units of power per unit length of wire.
Θ = temperature difference between wire and air.
v = linear velocity of air past wire.
c = a coefficient which includes a factor representing degree or nature of turbulence.

The air velocity is greater past the rear wire than past the front wire, because the radiator restricts the cross section of the stream, and

$$v_2 = \frac{v_1}{a} \tag{2}$$

where a = the fractional part of the frontal area of the radiator that is open for the passage of air, called its "free area," and the subscripts 1 and 2 represent the front and rear wires, respectively.

Turbulence at the front wire, i. e., in the open channel of the tunnel, is represented by the coefficient

$$c_1 = \frac{H}{\Theta_1\sqrt{v_1}} \tag{3}$$

and at the rear wire, i. e., in the radiator tube, by

$$c_2 = \frac{H}{\Theta_2\sqrt{v_2}} \tag{4}$$

The heat transfer H is the same in both cases, because the wires carry the same current and are of practically the same resistance per unit length. A comparison of turbulence in the radiator with that in front of it is obtained by substituting equation (2) in (4), and dividing by (3), which gives

$$\frac{c_2}{c_1} = \frac{\Theta_1}{\Theta_2}\sqrt{a}$$

As indicated above, the per cent of increase in resistance on being heated is used as a rough measure of the temperature difference. The values, including the ratio of the coefficients, are shown in Table III, and indicate differences in turbulence in different types of radiator, with the greatest turbulence in the perforated plate type, which whistles in an air stream.

TABLE III.—*Cooling of wires in the air stream.*

Type of radiator.	Air speed, m/sec.	Current, amperes.	Per cent increase in resistance.		Free area a.	Ratio of coefficients $\frac{\Theta_1}{\Theta_2}\sqrt{a}$.	Mean ratio.
			Front Θ_1.	Rear Θ_2.			
Flat plate	13	0.285	2.3	2.1	0.88	1.05	....
	13	.457	8.1	7.8	....	.98	....
	18	.456	7.4	7.0	....	1.00	1.01
Circular cell	13	.485	9.6	9.5	.65	.81	....
	17	.472	9.3	8.5	....	.88	....
	13	.304	3.7	3.3	....	.90	....
	17	.303	3.6	3.3	....	.88	....
	17	.497	9.9	9.0	....	.88	.87
Perforated plate	18	.208	0.8	0.5	.88	1.7	....
	18	.407	5.9	4.4	....	1.26	....
	13	.404	5.5	4.0	....	1.30	....
	13	.448	7.6	5.6	....	1.29	1.30

V. TEMPERATURE GRADIENTS.

In order to obtain some indication of the distribution of temperature within the air passages of radiators, the following procedure was followed:

A section of radiator was mounted in the 8-inch (20 cm) wind tunnel, and hot water was pumped through it as in calorimetric tests. A copper-constantan thermocouple was strung through one of the air tubes of the radiator, and supported by its own copper leads, in the manner described above for the platinum wires. The constantan wire was about 30 cm (12 inches) long, and the cold (upstream) junction was outside of the radiator in the stream of incoming air, when the hot (downstream) junction was in any position in the radiator tube, or even somewhat behind the rear face. Screw threads with a pitch of 0.16 cm ($\frac{1}{16}$ inch) on the rods supporting the wires, and on the supports for the rods, furnished rough micrometers for setting the position of the thermocouple and for moving it horizontally and vertically.

The mean temperature difference between the water in the radiator and the air passing through it, and the speed of the air stream, were maintained approximately constant, and corrections for variations in the temperature difference were made on the assumption that the temperature rise indicated by the thermocouple was proportional to this difference. No correction was made for slight fluctuations in the speed of the air stream, for trial showed that the effect on the thermocouple readings was small, even when the speed was varied over a wide range.

In order that the air might be disturbed as little as possible, fine wires were used, and were bared for some distance each side of the constantan section. At first No. 38 wire was tried, but so much trouble was experienced with breakage that most of the work was done with size No. 36.

Figures 2 to 5 show temperature gradients across the center of the tube in four radiators, at different distances from the front face, the sides of the plots representing the walls of the tubes. Figures 6 to 10 show isothermal lines plotted from the data shown on the other curves, and the upper and lower sides of the plots indicate the walls of the tubes. It must be emphasized that the quantities shown on the curves are very rough values, and can be used quantitatively only with very great caution, if at all, because the steepness of the temperature gradients across the tube, and the uncertainty in the position of a thermoelectric junction suspended by a meter of fine wire in an air stream make individual readings quite unreliable. Indeed, the uncertainty of position made it impossible to duplicate readings with any accuracy after the wires had been moved forward or backward, and readings were accordingly taken across the tube from one side to the other before moving the thermocouple to a new position along the stream. But although individual readings are not very reliable, the qualitative indications of the plots are probably correct.[7]

It has been suggested that at sections of the tubes near the forward end both viscous and turbulent flow might be found, each occupying a certain part of the cross section. If such were the case, a sudden break would be expected in the temperature gradient across the tube, at the boundary between the two kinds of flow, but no indication of such a break is found in the data; and although the inability to get reliable readings very close to the walls (because of contact between the swinging thermocouple wire and the wall) might have concealed this condition in some parts of the tube, it seems reasonable to suppose that it would have been detected at least in the longest tube.

The curves are of the form characteristic of turbulent flow, and are similar to curves of temperature gradients,[8] and isothermal lines [9] found by other observers when working with long tubes at velocities well above the critical values.

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[7] The possible effect of errors due to lead conduction in the thermocouple was investigated, and found to be entirely negligible in comparison with the known errors due to uncertainty of position.

[8] T. E. Stanton and Dorothy Marshall, British Adv. Com. Aero., Reports and Memoranda, No. 243, June, 1916.

[9] Groeber, Zeit. des Ver. Deut. Ing., 56, March, 1912, p. 421.

VI. EFFECTS OF TURBULENCE UPON RADIATOR PERFORMANCE.

In considering the effects of turbulence and turbulence devices on the performance of the radiator, it is necessary to bear in mind the importance of the *quantity* of air flowing through the core, and the fact that at a given speed *of flight* rather widely different amounts of air flow through different radiators. The comparisons made above have been based on a given air flow through the core, but speed of flight is the proper basis for comparing the general performance characteristics of a radiator.

Any form of construction that imparts additional turbulence to the air may be expected to increase the resistance to flow of air, and consequently to decrease the flow through the radiator for a given flying speed, while at the same time increasing the head resistance. If, then, there is to be a gain in general performance, any device for producing turbulence must, by increasing the amount of cooling surface, or by causing the air to scour the surface more thoroughly, or both, increase the heat transfer enough to overbalance both the decrease in amount of air flow (which tends to decrease the heat transfer), and the increase in head resistance.

The general performance of four types of radiator, each representing one of the best of its class, is shown in figure 11, by the "figure of merit," which is the ratio of the rate of dissipation of heat (expressed in units of power) to the power absorbed in overcoming the head resistance and sustaining the weight of the radiator.[10]

It is noticeable that at the higher speeds the flat plate and square cell types show much higher figure of merit than the other two types, although at a speed of 9 meters per second (20 miles per hour) the type with spiral vanes would perhaps be better than any of the others. The figures of merit as drawn apply only to radiators mounted in "unobstructed" positions, such that the flow of air through and around them is practically unaffected by other parts of the aircraft. For use in such positions at high speeds, every form of turbulence device known to this bureau is detrimental to the general performance of the radiator. On the other hand, if the radiator is to be used in such a position as the nose of the fuselage, the air flow through it at best is low, and an increase in air flow is accompanied by an increase in head resistance of the combination of fuselage and radiator. In this case, heat must be transmitted as rapidly as possible to the small amount of air that does flow through, and it may be profitable to use turbulence devices. It is possible that the rate of heat transfer for the whole radiator may be increased, while added air resistance of the core may actually reduce the head resistance of the fuselage and radiator.

[10] The figure of merit is computed on the assumption of temperature difference of 100° F. (55.6° C.) between air and water, and a "lift-drift" ratio of 5.4 for the airplane.

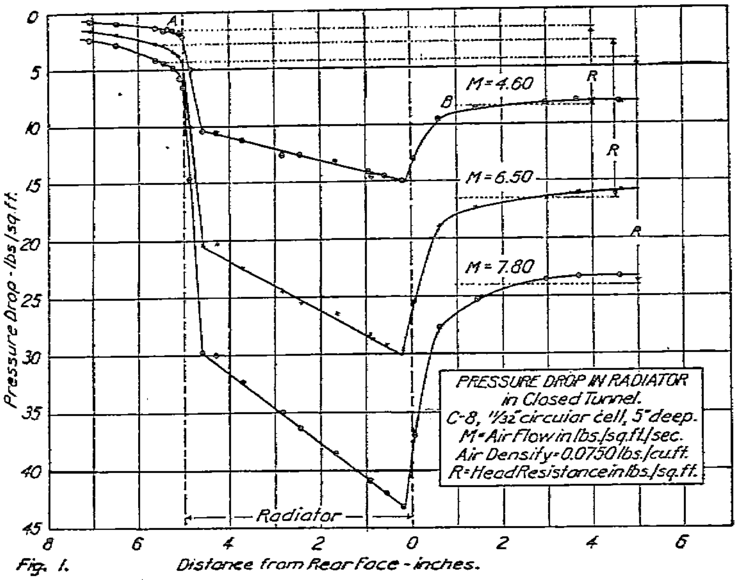

Fig. 1.

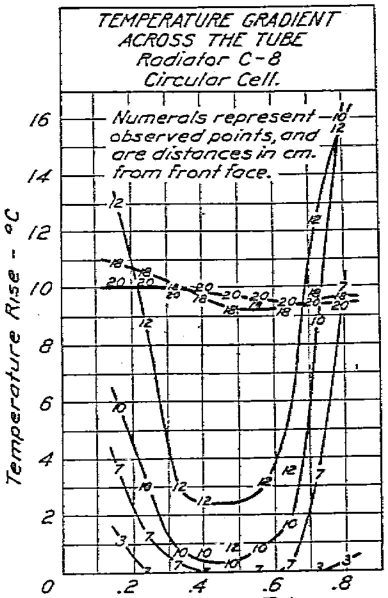

Fig. 2. Distance across Tube - cm.

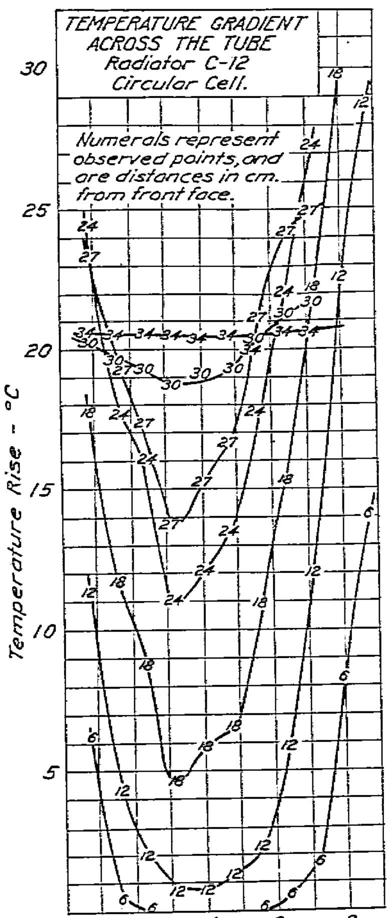

Fig. 3. Distance across Tube - cm.

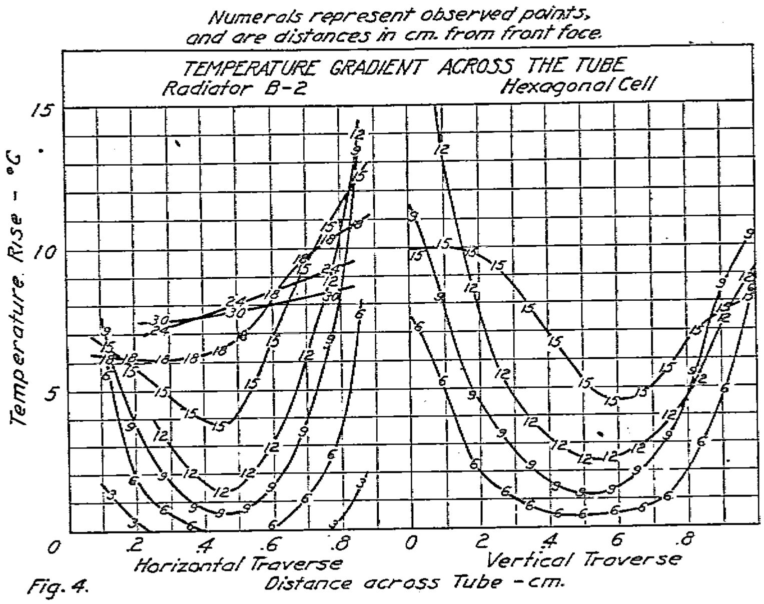

Fig. 4.

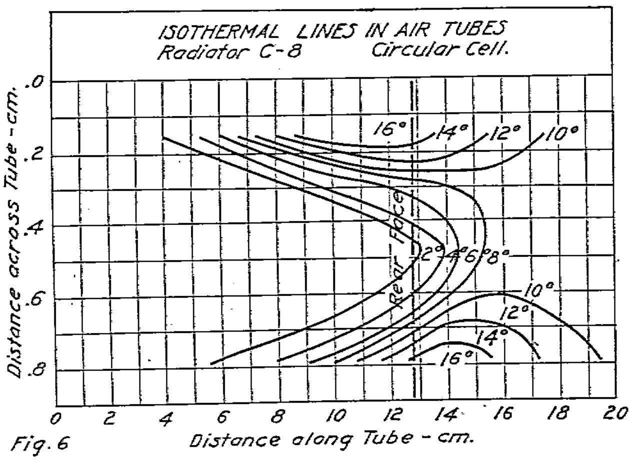

Fig. 6

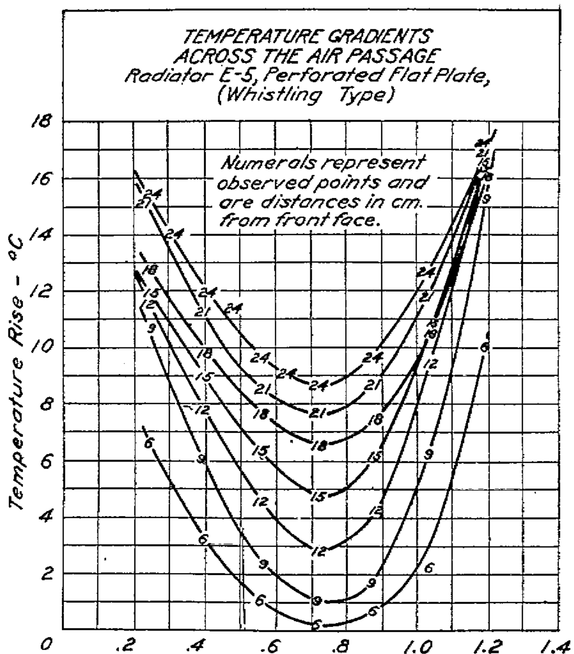

Fig. 5.

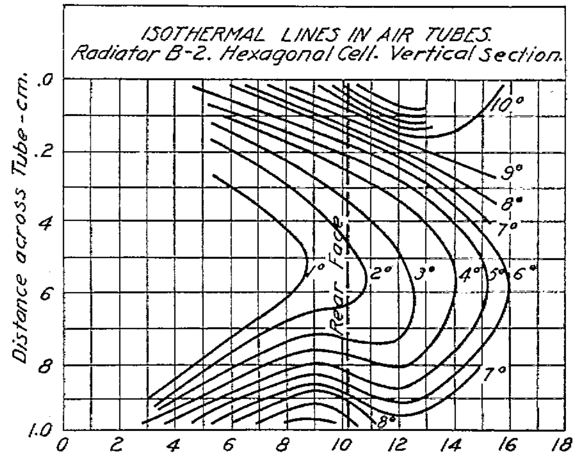

Fig. 9.

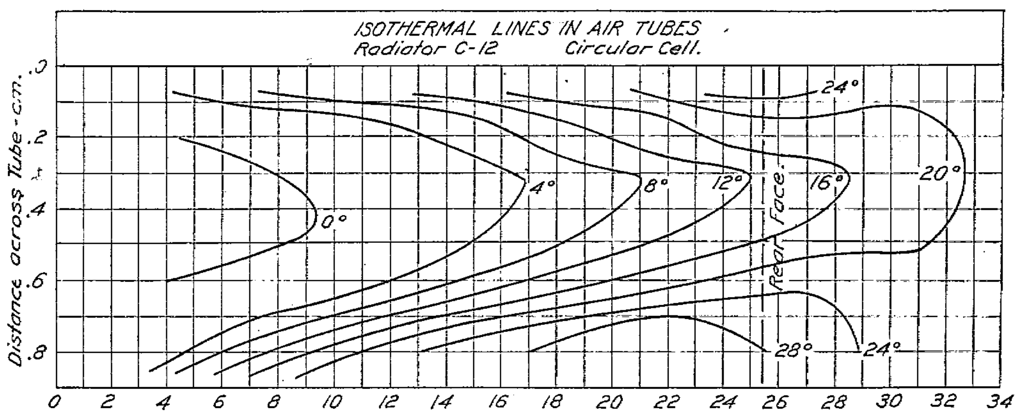

Fig. 7.

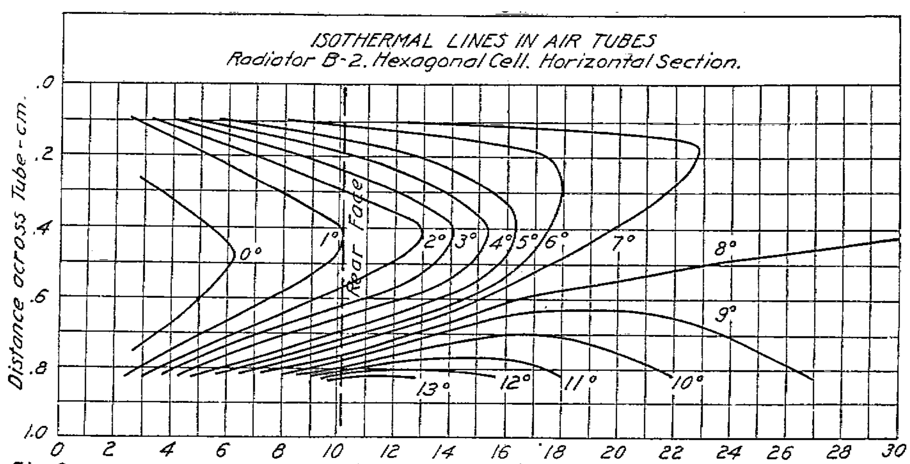

Fig. 8.

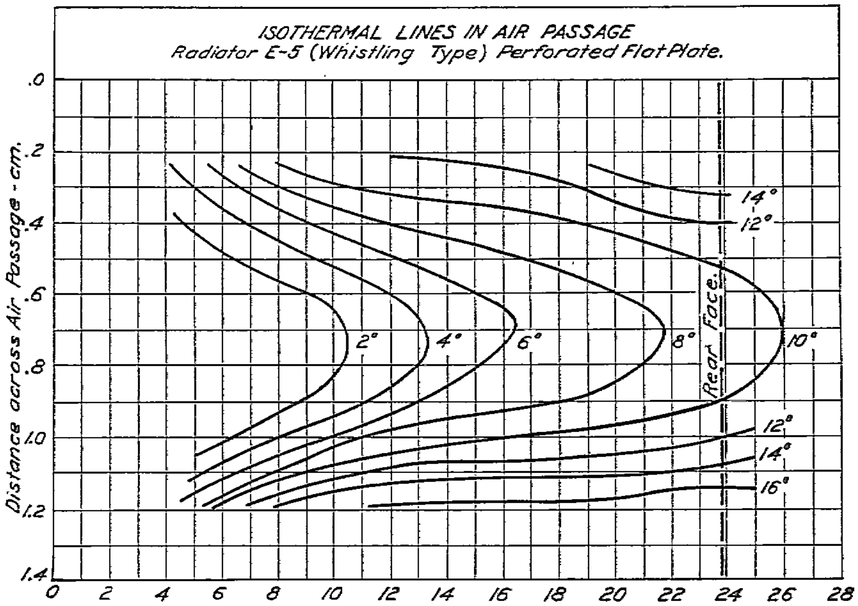
ISOTHERMAL LINES IN AIR PASSAGE
Radiator E-5 (Whistling Type) Perforated Flat Plate.
Distance across Air Passage - cm.
Distance along Air Passage - cm.
Rear Face.
2°
4°
6°
8°
10°
12°
14°
16°

Fig. 10.

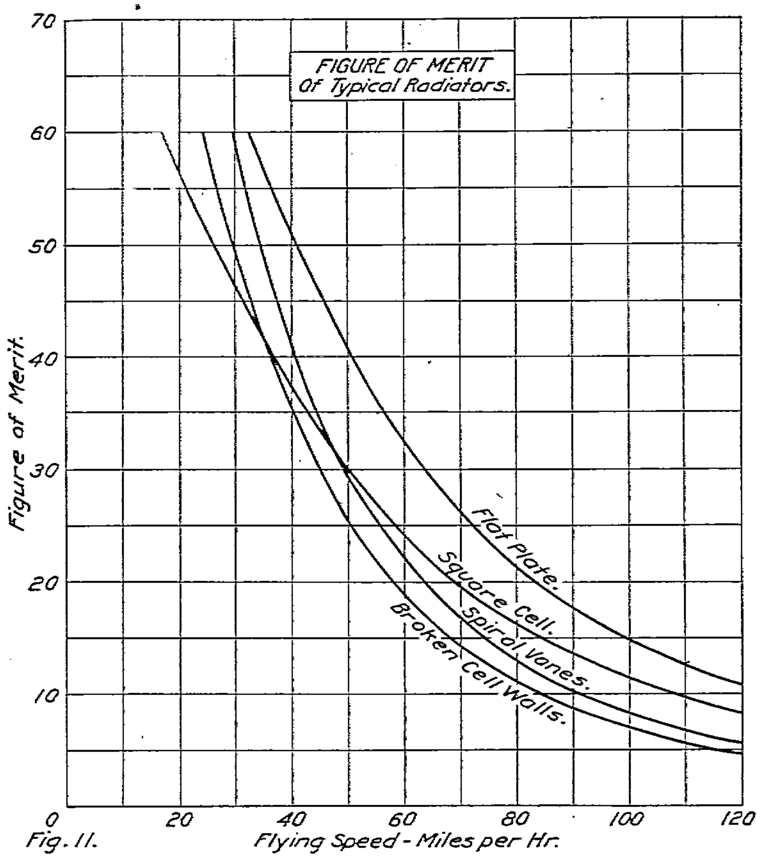
FIGURE OF MERIT
Of Typical Radiators.
Figure of Merit.
Flying Speed - Miles per Hr.
Flat Plate.
Square Cell.
Spiral Vanes.
Broken Cell Walls.

Fig. 11.